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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 5 :
F25B 41/00 // F25B 43/00
F25B 5/02

A1

(11) International Publication Number: WO 91/08427

(43) International Publication Date: 13 June 1991 (13.06.91)

(21) International Application Number: PCT/CA90/00420

(22) International Filing Date: 27 November 1990 (27.11.90)

(30) Priority data:
004,220 29 November 1989 (29.11.89) CA

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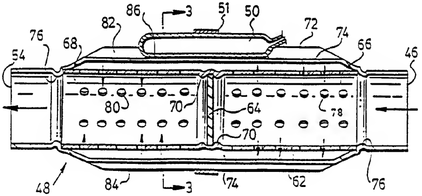
(81) Designated States: AT (European patent), AU, BE (European patent), BR, CA, CH (European patent), DE (European patent), DK (European patent), ES (European patent), FR (European patent), GB (European patent), GR (European patent), IT (European patent), JP, KR, LU (European patent), NL (European patent), NO, SE (European patent), SU.

Published
With international search report.

(54) Title: APPARATUS FOR THE SENSING OF REFRIGERANT TEMPERATURES



(57) Abstract

A device (48) is provided for sensing refrigerant temperatures in refrigerator systems, and for the control of refrigerant loading in a refrigerator evaporator circuit coil (34a, 34b). Such evaporator coils (34a, 34b) are supplied with refrigerant through a thermostatically controlled flow control valve (30) which is controlled by a sensor (50) to ensure a predetermined amount of superheat. The usual minimum superheat is about 5.5 °C (10 °F) and to reduce this value the refrigerant is passed through the device (48) in which it is rendered thoroughly turbulent and mixed, the device (48) intercepting the entire refrigerant flow just before the sensing of the superheat, thus ensuring that the temperature is accurately measured. The device (48) consists of a series of three chambers (66, 74, 68) connected together by two similar sets of holes (78, 80) the first and third chambers (66, 68) being similar so that it is completely reversible. The part of the device wall (72) intended to receive the sensor (50) may be provided with a groove (82) into which the sensor (50) fits snugly to increase the heat exchange contact between them. Grooves (82, 84) of different sizes can be provided in the same device (48) to receive sensors of respective different sizes. The heat conductive contact can be increased further by sandwiching a layer (86) of pasty heat conductive material between the sensor (50) and the groove wall (72) that fills the space between them.

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APPARATUS FOR THE SENSING OF REFRIGERANT TEMPERATURES

Field of the Invention

This invention is concerned with apparatus for the sensing of refrigerant temperatures in refrigerator systems and particularly with apparatus for the control of refrigerant loading in refrigerator evaporators.

Review of the Prior Art

The standard refrigeration compressor-operated system consists of a closed circuit in which cool low-pressure refrigerant vapor from a suction line enters a compressor which compresses it to a hot high pressure vapor, this hot vapor then flowing through a discharge line to a condenser coil or coils where it is cooled below its condensing temperature and becomes liquid. The liquid flows from the condenser through a return line into a liquid receiver, and from the receiver through a liquid line to an indicator and filter/drier, from whence it passes to a thermostatically controlled expansion valve which maintains at an optimum value the flow of the liquid refrigerant into an evaporator coil or coils, in which it evaporates with consequent temperature drop and cooling of the coils and their environment; the resultant vapor passes through the suction line back to the compressor to complete the circuit.

It is essential to control the expansion valve (usually called the TX valve) so as to prevent any liquid refrigerant from reaching the compressor, which would damage it, and this valve control usually consists of a remote temperature sensing fluid-containing cylindrical bulb connected by a metal capillary tube to a charged diaphragm capsule in the valve. The capsule responds to changes in temperature of the sensor bulb to regulate the flow through the valve. Equivalent electrical sensors have also been developed. The sensor bulb or its equivalent normally is clamped tightly to the suction line at the exit from an outlet manifold into which the evaporator coil or group of coils discharge, so as to sense the temperature of the vapor at this point. The temperature characteristic of a

vaporizing body of liquid is very standard in that its temperature will remain relatively constant at about the respective vaporizing (saturation) temperature as long as there is some liquid present to vaporize, and then will rise relatively rapidly when all the liquid is gone. To ensure that no liquid escapes from the evaporator the sensor is set for an operating temperature sufficiently higher than the saturation temperature, and the difference between these two temperatures is known as the superheat. As an example, a quite usual range of values for the saturation temperature of such a system is about -7°C to about 4.5°C (20°F to 40°F), while a quite usual value for the superheat is about 5.5°C (10°F), so that the range of control temperatures for such systems will be -1°C to 10°C (30°F to 50°F).

In theory it should be possible to use a much lower superheat value, say 1°C (2°F), but in prior art practice it was found that this was not sufficient to ensure the complete absence of liquid refrigerant from the evaporator manifold outlet and the higher value was therefore almost universally used. As the superheat value varies around the predetermined amount the TX valve opens and closes, and in theory should be operable to maintain it quite accurately at that value, but in practice there is a time lag between the sensing of the temperature by the sensor and the operation of the TX valve, which also usually cannot respond fast enough, resulting in a fluctuating superheat value necessitating the higher amount, thereby reducing the efficiency of the system. There has therefore been a continuing need for a temperature sensor for such systems which can more accurately determine the temperature of the refrigerant vapor in the suction line and thus improve the efficiency.

In commercial refrigerators, most evaporators consist of a large number, often as many as fifty, separate "circuit coils" connected in parallel so as to obtain sufficient cooling capacity without the individual coils being of too great length with consequent high pressure drop. These circuit coils are arranged in sets, each set having its own expansion valve and a common distributor interposed between the valve and the coils of

the set, the purpose of the distributor being to divide the flow as equally as possible between individual small diameter feed pipes of equal length leading from the distributor to the respective circuit coil pipe inlets. All of the circuit coil pipe outlets are connected to a common outlet manifold or stand-pipe. Despite the care that is taken to try to make the valve and the distributor feed equal amounts of liquid refrigerant to the circuit coils, and to make all of the circuit coils as equal in length and flow characteristic as possible, it is in practice always found that liquid refrigerant vaporizes in some of the coils at a different rate than in the others, due to variables such as differences in the flow of air over the different coils, and small differences in the pressure drop through each coil. The consequence is that the circuit coil or coils which absorb the least amount of ambient heat allow the liquid refrigerant to flow further along it or them before vaporizing, so that it is this coil or coils that control the TX valve and close it down, starving the remainder of the coils of liquid refrigerant and excessively superheating the refrigerant vapor in the starved coils, and thereby reducing the cooling capacity of the system. This reduction can be as much as from about 25 to 35% of the total capacity.

This unequal loading of the evaporator circuit coils can usually be observed by visual inspection of the coils once the system has been in operation of a short time, when the starved circuit coils are less frost coated toward the outlet end than the others. This unequal loading is often mistakenly attributed to unequal distribution of the refrigerant liquid among the coils.

Definition of the Invention

It is therefore a principal object of the present invention to provide a new apparatus for the sensing of refrigerant temperatures in refrigerator systems, and in particular a new apparatus by which the temperature of the refrigerant exiting from an evaporator coil is sensed efficiently by the temperature sensor controlling the TX valve for more precise superheat control.

In accordance with the present invention there is provided apparatus for the sensing of the temperature of refrigerant exiting from a refrigeration system evaporator coil outlet and for the control in accordance with the temperature sensed by a sensing means of a controllable evaporator valve feeding liquid refrigerant to the evaporator coil inlet, the apparatus comprising:

a turbulating and mixing device having an inlet and an outlet for refrigerant and having therein a refrigerant flow path having at least part of a wall thereof of heat conductive material for sensing the enclosure device interior temperature through the wall part;

the device comprising a first tubular member having at least approximately midway along its interior a transverse barrier dividing the interior into a first chamber connected to the inlet and a second chamber connected to the outlet and against which the refrigerant flow impinges to produce resultant turbulence in the first chamber;

a second tubular member surrounding the first tubular member to form an annular second chamber between them;

a first set of bores provided in the wall of the first passage and directing the refrigerant flow into the second chamber against the inner wall of the second tubular member, and;

a second set of bores provided in the wall of the third chamber directing the refrigerant flow from the second chamber into the third chamber.

Description of the Drawings

Embodiments of the invention will now be described, by way of example, with reference to the accompanying schematic and diagrammatic drawings, wherein:

Figure 1 is a schematic diagram illustrating a typical refrigeration system and including a device that is a first embodiment of the invention;

Figure 2 is a longitudinal cross-section to a larger scale of the device of Figure 1;

Figure 3 is a transverse cross-section of the device of Figure 2, taken on the line 3-3 in Figure 2; and

Figure 4 is a transverse cross-section similar to Figure 3 through another device of the invention.

Description of the Preferred Embodiments

Referring now to Figure 1, a typical refrigeration system to which the apparatus of the invention can be applied comprises a refrigerant compressor 10 having a suction inlet 12 and a high pressure outlet 14, the compressor feeding the hot compressed refrigerant fluid via conduit 15 to a condenser coil 16 having an inlet 18 and an outlet 20. Cooled refrigerant from the coil 16 passes via conduit 21 to a liquid accumulator 22, and thence via conduit 24 through a filter/drier 26, a liquid indicator 28 and a common thermostatically controlled refrigerant flow control TX valve 30 into a distributor 32, from which it flows into two parallel-connected circuit coils 34a and 34b of an evaporator coil. For convenience in illustration only two circuit coils are shown, but in practice there can be as many as fifty in a single large evaporator coil, each circuit coil being connected by a respective inlet pipe 36a and 36b to the common distributor 32. As described above in practice care is taken to make all of the circuit coils 34a, 34b, etc., and all of the pipes 36c, 36b, etc., of the same length and as equal as possible, so that the refrigerant will be distributed as equally as possible among them.

Each circuit coil has an inlet 38a, 38b respectively and an outlet 40a and 40b respectively, the latter all being connected to a common header pipe 42 (sometimes also called a stand-pipe or manifold), the single outlet 44 of which is connected to inlet 46 of a turbulator and mixing device 48 of the invention. A superheat temperature sensing bulb 50 by which the TX valve 30 is controlled is tightly clamped to the exterior of the device 48 by a clamp 51 to be in good heat exchange with its interior through the device wall and is connected by a capillary tube 52 to the valve 30. The outlet 54 of the device 48 is connected by conduit 56 to the pump inlet 12 to complete the system circuit. The usual fans 58 and 60 are provided to circulate ambient air over the coils 16 and 34a, 34b respectively. The numerous other circuit elements, controls and

indicating devices that such a system normally includes do not constitute part of this invention and therefore do not need to be illustrated. The direction of flow of the refrigerent is indicated by the broken arrows.

Referring now also to Figure 2, this particular device 48 is made of metal, preferably a high conductivity metal such as copper or brass, and consists of a first inner cylindrical pipe 62, provided at least approximately at its middle along its length with a transversely-extending circular disc 64 comprising an end barrier extending over its entire cross-sectional area and dividing the interior of the pipe into two separate independent cylindrical chambers 66 and 68, called for convenience in terminology the first and third chambers. In this embodiment the disc is retained in position by its entrapment between two radially inwardly extending circular ridges 70 produced by a die-forming operation in the pipe; it may be noted that the joint between the disc and the inner wall of the pipe does not need to be absolutely gas tight. A second outer cylindrical pipe 72 having a central portion of larger diameter than its two end portions surrounds the first inner pipe 62 coaxial therewith and is sealed to the pipe at both ends, thereby forming an annular cross-section second chamber 74 between the two pipes. The inner pipe is held securely within the outer pipe between two radially inwardly extending circular ridges 76 die-formed in the outer pipe, and the ends of the outer pipe are reduced in diameter to the size required for the system in which it is inserted, one end constituting the inlet 46 while the other end constitutes the outlet 54.

The fast flowing refrigerant fluid entering the pipe 62 impinges strongly against the transverse barrier 64 and immediately becomes extremely turbulent within the first chamber 66. The inner pipe has a first set of a plurality of holes 78 distributed uniformly along the part of its length within the first chamber 66, and also distributed uniformly around its periphery, which holes direct the turbulent refrigerant vapor in the chamber 66, together with any liquid entrained therein, forcibly into the chamber 74 against the inner wall of the outer pipe 72. The inner pipe has another set of a plurality of holes

80 similarly uniformly distributed along the part of its length within the second chamber 68 and around its periphery, which holes direct the highly turbulent vapor in the chamber 74 back into the third chamber 68 and out of the outlet 54, the abrupt change of direction of the vapor required for its passage through the second set of holes 80 considerably increasing its turbulence in the chamber 74. The pipes 62 and 72, the barrier 64, and the bores 78 and 80 therefore provide within the interior of the device a direction-changing flow path between the inlet and the outlet that produces a thorough turbulating and mixing action on the refrigerant. The combination of the tortuous path formed by the successive chambers 66, 74, and 68, the abrupt changes in direction of the fast-flowing fluid, the turbulence in the inner pipe chamber 66 because of the impingement of the fluid against the closed end, and the turbulence in the annular chamber 74 between the two pipes because of the said impingement against the outer pipe inner wall, and its subsequent change of direction to exit through the holes 80, all ensuring that the entire refrigerant flow in the flow path, whether in the liquid or vapor phase, is all thoroughly mixed and rendered turbulent, and particularly without any possibility of the relatively high velocity vapor phase being able to flow through the device separately from the liquid phase. Moreover, the vigorous impingement of the high velocity fluid against the outer pipe inner wall ensures that any relatively stagnant barrier layer of refrigerant, or of the lubricating oil that is always entrained therein, is thoroughly disrupted and removed from the inner wall, so that it cannot prevent the efficient transfer of heat from the refrigerant through the wall to the sensor bulb 50. The bulb is therefore sensing only the temperature of a completely turbulent mixed and temperature averaged refrigerant flow as received from the outlet of the header pipe 42, and in addition is much more sensitive to changes in the refrigerant temperature and more accurately measures the device interior temperature which corresponds to the averaged refrigerant temperature. This turbulating and mixing function of the device 48 is effective in this manner whatever the evaporator coil structure employed in

the system.

Unexpectedly I have found that a device as specifically described, employing three successive chambers with two abrupt changes of direction through respective sets of holes, is more efficient in providing for accurate measurement of the temperature of the fluid refrigerant than my prior device, as described and illustrated for example in Figure 2 of my prior U.S. Patent No. , which employs two successive chambers with only a single abrupt change of direction through a single set of holes. Another substantial commercial advantage of this embodiment is that it is symmetrical end for end and completely reversible, so that it is immaterial which end is employed as the inlet and which is employed as the outlet; the installer is therefore able to instal it in the line 56 without having to consider the direction of refrigerant flow through the device. It was found with the prior art devices that there was a small but noticeable decrease in performance if it had been installed reversed, but this cannot happen with the devices of the present invention.

Another improvement in performance is obtained in apparatus in accordance with this invention by providing the outer pipe 72 with a longitudinal exterior groove 82 of cross-section corresponding to that of the sensor bulb 50, so that there is maximum practical heat-exchange surface contact between the bulb and the pipe 72, and thus between the interior of the chamber 74 and the bulb. In practice sensor bulbs as used in refrigeration systems are almost universally of cylindrical shape and circular cross-section, and accordingly the groove 82 is made of semi-circular cross-section of size such that the bulb will fit snugly within the groove and be held firmly in place by the band clamp 51. It is also found in commercial practice that sensors are usually either of diameter 12.8 mm (0.5 in) or 9.5 mm (0.375 in), and accordingly the tube 72 is provided with two circumferentially spaced grooves 82 and 84 of these two different diameters, so that the installer can chose the one appropriate for the size of bulb to be used. More than two spaced grooves can be provided if more than two sizes are involved. The formation of the groove or grooves, usually

by a die-forming operation, will result in a small decrease in the cross-section area of the annular passage 74, and this can readily be compensated, if required, by a small increase in diameter of the central portion of the pipe 72. Neither the pipe 72 nor the bulb 50 are likely in practice to be manufactured to close tolerances, and to ensure even better heat exchange contact between them the wall of the selected groove may be pre-coated with a layer 86 of heat-conductive paste, which is squeezed between them as the bulb is pressed into the groove by the band clamp 51 and fills any air space that might otherwise be left between them.

In prior practice the circular cross-section bulb has usually been clamped to the exterior of the circular cross-section pipe with their axes parallel, and at best there is only a line contact between them where their peripheries touch one another. In practice the situation is often much worse in that, if the installer does not mount the bulb carefully it may be somewhat skewed relative to the pipe, with the axes no longer parallel, whereupon the area of contact is correspondingly reduced. This can very easily happen since it has been usual to attach the bulb by means of two longitudinally-spaced band clamps, and it is comparatively easy during the installation of the second clamp for the bulb to become skewed; and even a small amount of skew causes a considerable reduction in contact area. With a device of this aspect of the invention all that is required is for the installer to select which of the grooves is closest in size to the bulb while able to receive it, apply a bead or layer of the heat conductive paste material to the wall of the groove, place the bulb in position in the groove and clamp it firmly in place using only a single clamp, whereupon accurate positioning and alignment is immediately assured.

The grooves 82 and 84 can be made of cross-sections that are more than semi-circular to increase the contact area, but the bulbs must then be slid endwise into the groove, which may be difficult in some installations; such re-entrant grooves are more difficult to manufacture than the open-mouth semi-circular grooves illustrated.

The devices of the invention have the advantages both to the installer, and to the owners of the apparatus in which they are installed, that they not only produce an improvement in performance of the system by permitting a substantial reduction in the superheat, but they provide a pre-established preferred and easier installation location for the sensor bulb that both ensures the improved performance will be obtained and also simplifies the installation procedure. Thus, in practice once the decision has been made to install a device of the invention, which is easily done with new equipment, and which for retrofitting to old equipment involves the relatively simple procedure of cutting a section from the pipe to permit its insertion therein, the hitherto sometimes difficult questions of proper location and correct installation of the sensor bulb are also readily and positively determined.

When the device is used with a system as specifically described, namely with multiple circuit coils, then in addition to turbulating and mixing the fluid flow in each evaporator circuit coil it also performs a multiple mixing function, whereby the fluid flows from all of the circuit coils are thoroughly mixed together, so that all of their separate temperatures are averaged, and it is this average circuit coil temperature that is detected by the bulb 50. Moreover, this very thorough turbulence and mixing ensures that if one or more of the circuit coils is not evaporating all of its supply of refrigerant, then the small quantities of liquid reaching the mixing device are immediately atomized and consequently easily vaporized by heat from the superheated vapor from the remaining coils. The supply of refrigerant to the starved coil or coils can therefore be increased until the superheated vapor they produce is not able to vaporise the liquid refrigerant from the underloaded coil or coils.

The diameters of the pipes 62 and 72 are such that the flow capacities of the resultant flow passages are about that of the remainder of the suction tube 56, while the number and size of the apertures 78 and 80 are such that about the same flow capacity is achieved. These flow capacities can vary between about 0.5 and 1.5 times the usual flow capacity of the suction

tube; it may be preferred to reduce the flow capacity of the apertures 78 somewhat below that of the apertures 80 and that of the suction tube in order to obtain sufficiently forceful impingement of the fluid against the outer tube inner wall.

In one specific embodiment intended for use in a system of about 2-3 h.p. the outer pipe 72 is about 23 cm (9 ins.) long and 3.5 cm. (1.375 ins.) maximum outside diameter; the inner pipe 62 is about 17 cm (6.75 ins.) long and 2.2 cm (0.875 in.) inside diameter and is provided with the two sets of uniformly spaced holes, each of which is 3.1 mm (0.125 in.) in diameter. Each set consists of six circumferentically-spaced rows, each of seven holes, for a total of forty·two holes for each set.

In another specific embodiment intended for use in a system of about 10-15 h.p. the outer pipe 72 is about 23 cm (9 ins.) long and 6.35 cm. (2.5 ins.) maximum outside diameter; the inner pipe 62 is about 17 cm (6.75 ins.) long and 4.4 cm (1.75 in.) inside diameter and is provided with the two sets of uniformly spaced holes, each of which is 6.3 mm (0.25 in.) in diameter. Each set consists of five circumferentically-spaced rows, each of six holes, for a total of thirty holes for each set.

In practice if the system is of power from about 40 H.P. and up it is usual to split the coils into two sets and to provide each with a separate TX valve controlled from its respective sensor. Except therefore for special installations it is unlikely that a device with an inner pipe 62 of more than 7.8 cm (3.125 ins) O.D. will be required. It is found in practice that the pressure drop through the devices of the invention is sufficiently low, usually less than about 1 p.s.i., that it does not produce any appreciable loss of efficiency, and any loss for this reason is amply compensated by the overall considerably improved efficiencies that usually are obtained. The drop is sufficiently small that it is difficult, if not impossible, to detect with the pressure gauges that are used in standard refrigeration service practice.

Despite the lengthy period of time for which these problems have existed it does not appear to have been understood how to provide turbulator means and/or mixing means that will

sufficiently improve the temperature detection and control of the TX valve, and also in multiple coil systems to average the temperatures of the refrigerant flows from the large number of individual circuit coils for the same purpose. Thus, the current commercial literature in the industry of which I am aware seems to assume that all that can be done is to make the lengths of the circuit coils as equal as possible, to discharge all of the circuit coils into a common header pipe, and to clamp the sensor bulb to the outside of the outlet pipe from the header pipe, when the temperature will be measured as accurately as possible and the flows will be mixed to the maximum obtainable extent.

I believe that this mistaken assumption may have resulted from a lack of adequate appreciation of the flow conditions of the refrigerant fluid in the evaporator coils and the outlet pipe or manifold. The refrigerant enters the coils as a low volume liquid and is evaporated in the confined spaces thereof to a high volume vapor, with the result that the exit speed of the vapor is relatively high, to the extent that in the absence of the highly positive turbulating and/or mixing apparatus of the invention, involving the entire fluid flow or flows, the flows in the coils remain laminar and any liquid particles remain entrained without mixing, while there is little or no opportunity for the flows from the different coils to mix and average. Consequently there is little opportunity for any small quantities of liquid refrigerant to be evaporated before the temperature must be sensed by the bulb 50. It is essential for the turbulating and mixing to be carried out across the entire cross-section of the flow path, since any gaps will allow the corresponding portion or portions of the high velocity fluid passing through them to remain laminar with liquid particles entrained and defeat the purpose of the device. The situation would not be made much better in the prior art apparatus by placing the sensor bulb 50 further along the suction pipe 56, since the flows will still remain relatively laminar along the pipe, and any additional distance of the bulb from the evaporator outlet and from the TX valve introduces additional difficulty because of the increased time delay for operation of

the TX valve.

As evidence of this current lack of appreciation of the problem there is and has been considerable discussion of the best physical arrangement for the coils to ensure that they are equally loaded, and it has been considered important in prior refrigeration systems to locate the sensor bulb 50 appropriately on the circumference of the suction pipe in order to sense the superheat temperature as accurately as possible and operate with minimum superheat. The manufacturers of TX valves in their installation manuals stress the importance of proper location of the sensor bulb, but do not give a definitive location for it. They advise that preferably the bulb should be fastened to a horizontal portion of the suction line, and clamped at different places around its circumference depending on the diameter, but the location is finally chosen by the installer depending upon what appears to be suitable and/or practicable for that installation, often with poor results. The theoretically ideal location is at 6 o'clock on the circumference of a horizontal suction pipe, where it should be able to sense most accurately any small quantity of liquid refrigerant passing in the pipe, and would therefore permit the smallest amount of superheat. In practice this has not been a satisfactory location because of the presence of lubricant oil in the refrigerant, which flows along the bottom of the pipe and would thermally insulate the sensor bulb from the refrigerant fluid. The usual location for the bulb has therefore been at four or eight o'clock on the pipe circumference. It is found that with the thorough turbulence and mixing provided by the devices of the invention the location of the sensor bulb around the circumference of the device is no longer critical, and it can be placed at the most convenient location from the point of view of installation and subsequent access for service. In addition it is now found unnecessary to locate the sensor bulb on a horizontal portion of the suction line, and the attitude of the device has no effect upon its performance.

The effectiveness of a device of the invention can readily be seen by visual inspection of the evaporator coil before and after its installation. Before installation it is

usually found that the frost deposition on the different circuit coils is non-uniform with some of them completely frosted up to the outlet, while others are not frosted for a substantial distance back from the outlet, showing that the latter are starved of refrigerant and are working much below their maximum cooling capacity. Also the evaporator common outlet member is only partially frosted. With the device installed all of the circuit coils become more or less equally frosted, as well as the entire length of the suction manifold, indicating that all of the circuit coils are now operating at their full designed capacity. It is now found possible safely to reduce the amount of superheat from the prior value of about 5.5°C (10°F) to as low as 2°C (4°F). In some installations the resultant improvement in cooling capacity of the system can reach as much as 25-35%, indicating that the system previously was operating at only 74-80% of the available capacity.

As a specific example, in an installation employing compressors totaling 200 H.P. and eight forced air evaporator coils the system prior to the installation of the devices of the invention took 3 hours, 10 minutes to cool the room temperature from 13°C (55°F) to -19°C (-2°F). With the devices installed the time taken was reduced to 2 hours, 10 minutes, an improvement of 29% in efficiency or equivalent to increasing the output of the compressors to about 258 H.P.

An important advantage that has been found to follow from use of the invention, demonstrating its unexpected nature, is the flexibility that is obtained upon installation in not having to closely match the size of the TX valve to the evaporator coil capacity without the valve losing control of the refrigerant flow. The capacity of a TX valve is determined both by the size of its flow aperture and the head pressure across the aperture, and it has been important in prior art installations for this match to be as close as possible. For example, one manufacturer provides 21 different sizes of valve to cover the range 0.5-180 tons, those in the range 0.5 - 3 tons being rated in 0.5 ton increments, with progressively increasing intervals up to the maximum. If the valve is too large then with the high superheat values employed the valve hunts,

overfeeding and underfeeding the evaporator with resultant poor efficiency and danger of liquid reaching the compressor because of the over-large flow capacity of the valve while open. On the other hand, with the valve and coil sizes closely matched it becomes necessary to maintain the head pressure above a minimum value, since otherwise the valve flow capacity becomes too low. This penalizes the system in winter when the air cooled condensers are very efficient and could operate with lower head pressure; instead it is necessary to maintain it artifically high by various techniques that are available. This means that the power required to compress the refrigerant must also be maintained at a corresponding high uneconomical value.

This loss of control is easily observed in practice. For example, if the evaporator fan stops for some reason, perhaps a broken fuse, or the flow of product being cooled is interrupted, the load on the coil drops suddenly, faster than can be controlled by the valve, and liquid floods the compressor, which then becomes covered with frost when it should be frost-free. The liquid refrigerant washes out the lubricant, and can cause valve breakage and damage. Again, if the automatic coil defrost system is not operating satisfactorily and the coils become coated with ice the load on each coil drops and control can be lost; this of course is easily detected by visual inspection of the coils.

Upon installation of a device or devices of the invention it is found that this close match of load capacities is no longer necessary and an oversize valve can be employed successfully. In a specific example, in a system with a 1.5 ton evaporator the original 2 ton rated valve was replaced with an 8 ton rated valve; adequate control was maintained with the superheat value fluctuating about 0.5°-1°C (1°- 2°F). Thus with a larger orifice TX valve it is no longer necessary to keep the head pressure at an artificially high value to maintain adequate refrigerant flow through the valve, and instead it could be allowed to drop to a lower level and still maintain proper superheat control with maximum evaporator capacity. This not only maximizes the efficiency of the system but also provides the possibility of reducing the number of different sizes of

valves required for a full range of installation sizes.

In commercial refrigeration practice circular cross-section pipes are universally used, and pipes of such cross-section are illustrated for the device shown in Figures 1-3. However, the devices of the invention can also be made using pipes of other cross-sections, such as the square cross-section illustrated by Figure 4.

I CLAIM

1. Apparatus for the sensing of the temperature of refrigerant exiting from a refrigeration system evaporator coil outlet (40a, 40b) and for the control in accordance with the temperature sensed by a sensing means (50) of a controllable evaporator valve (30) feeding liquid refrigerant to the evaporator coil inlet (38a, 38b), the apparatus comprising:

a turbulating and mixing device (48) having an inlet (46) and an outlet (54) for refrigerant and having therein a refrigerant flow path (46, 66, 78, 80, 68, 54) having at least part of a wall (72) thereof of heat conductive material for sensing the device interior temperature through the wall part;

the device (48) comprising a first tubular member (62) having at least approximately midway along its interior a transverse barrier (64) dividing the interior into a first chamber (66) connected to the inlet (46) and a second chamber (68) connected to the outlet (54) and against which the refrigerant flow impinges to produce resultant turbulence in the first chamber (66);

a second tubular member (72) surrounding the first tubular member (62) to form an annular second chamber (74) between them;

a first set of bores (78) provided in the wall of the first chamber (66) and directing the refrigerant flow into the second chamber (74) against the inner wall of the second tubular member (72), and;

a second set of bores (80) provided in the wall of the third chamber (68) directing the refrigerant flow from the second chamber (74) into the third chamber (68).

2. Apparatus as claimed in claim 1, wherein the transverse barrier (64) is retained in the first tubular member (62) interior by radially inwardly extending parts (70) of the first member wall.

3. Apparatus as claimed in claim 1 or 2, wherein the first

tubular member (62) is retained inside the second tubular member (72) by radially inwardly extending parts (76) of the second member wall.

4. Apparatus as claimed in any one of claims 1 to 3, wherein a part of the wall of the second tubular member (72) is provided with a grooved portion (82 or 84) in which the sensing means (50) can be disposed, the cross-section of the grooved portion corresponding to the cross-section of the sensing means to increase the heat conductive contact of the sensing means with the wall portion.

5. Apparatus as claimed in claim 4, wherein the sensing means (50) comprises an elongated tubular member and the grooved portion (82 or 84) has a longitudinal axis extending parallel to the longitudinal axis of the second tubular member (72).

6. Apparatus as claimed in claim 4 or 5, wherein the wall of the second tubular member (72) is provided with at least two circumferentially spaced grooved portions (82 and 84) of different cross-section sizes for receipt of sensing means (50) of respective cross-section sizes.

7. Apparatus as claimed in any one of claims 4 to 6, wherein the wall of the grooved portion (82 or 84) contacted by the sensing means (50) is provided with a layer of heat conductive paste (86) that is engaged by the sensing means to improve heat conduction from the wall to the sensing means.

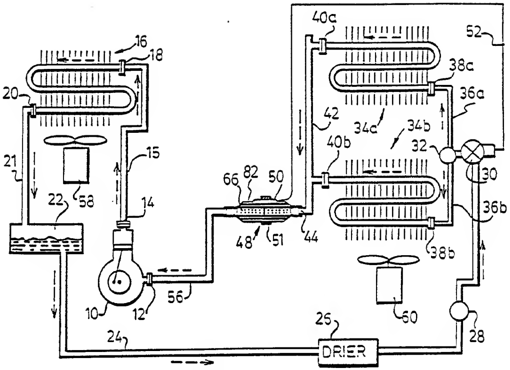

FIG.1

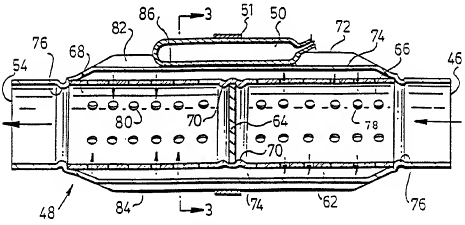

FIG.2

SUBSTITUTE SHEET

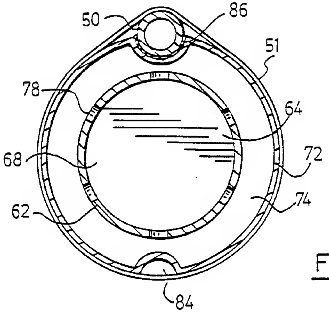

FIG.3

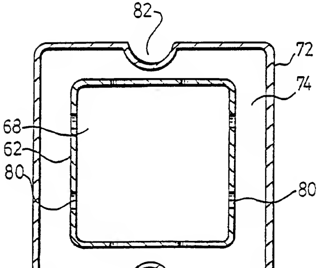

FIG.4

SUBSTITUTE SHEET

INTERNATIONAL SEARCH REPORT

International Application No PCT/CA 90/00420

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all)[6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int.Cl. 5 F25B41/00 ; //F25B43/00 ;//F25B5/02

II. FIELDS SEARCHED

Minimum Documentation Searched[7]

Classification System	Classification Symbols
Int.Cl. 5	F25B ; G01K ; G05D

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched[8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT[9]

Category °	Citation of Document,[11] with indication, where appropriate, of the relevant passages[12]	Relevant to Claim No.[13]
A	GB,A,2014290 (STAL REFRIGERATION) 22 August 1979 see page 1, line 116 - page 2, line 26; figures 1, 3 ---	1
A	DE,C,722412 (BROWN,BOVERI&CIE) 09 July 1942 see page 2, line 111 - page 3, line 47; figure ---	1, 4, 5
A	US,A,4694894 (KITO ET AL.) 22 September 1987 see the whole document ---	1, 4, 5
A	FR,A,1178599 (SULZER) 12 May 1959 see the whole document ---	1, 4, 5, 7
A	DE,B,1068734 (GOEBELS) 12 November 1959 see column 1, line 51 - column 2, line 49; figures 1, 2 ---	1, 5
P,A	EP,A,354037 (GREGORY) 07 February 1990 see the whole document ---	1
	-/--	

° Special categories of cited documents :[10]

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"P" document published prior to the international filing date but later than the priority date claimed

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"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
21 FEBRUARY 1991	19. 03. 91
International Searching Authority	**Signature of Authorized Officer**
EUROPEAN PATENT OFFICE	BROMAN B.T.

III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)

Category °	Citation of Document, with indication, where appropriate, of the relevant passages	Relevant to Claim No.
A	US,A,4798058 (GREGORY) 17 January 1989 ---	

ANNEX TO THE INTERNATIONAL SEARCH REPORT ON INTERNATIONAL PATENT APPLICATION NO. PCT/CA 90/00420

SA 42085

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report.
The members are as contained in the European Patent Office EDP file on 21/02/91
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

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		FR-A,B 2416432	31-08-79
		SE-A- 7801379	08-08-79
		US-A- 4229949	28-10-80
DE-C-722412		None	
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		JP-A- 61114093	31-05-86
FR-A-1178599		None	
DE-B-1068734		None	
EP-A-354037	07-02-90	JP-A- 2238255	20-09-90
US-A-4798058	17-01-89	None	

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82